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FRICTION OF COMPRESSION-IGNITION ENGINES

By Charles S. Moore and John H. Collins, Jr.
Langley Memorial Aeronautical Laboratory

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SUMMARY

The cost in mean effective pressure of generating air flow in the combustion chambers of single-cylinder compression-ignition engines was determined for the prechamber and the displacer-piston types of combustion chamber. For each type a wide range of air-flow quantities, speeds, and boost pressures was investigated. Supplementary tests were made to determine the effect of lubricating-oil temperature, cooling-water temperature, and compression ratio on the friction mean effective pressure of the single-cylinder test engines. Friction curves are included for two 9-cylinder, radial, compression-ignition aircraft engines.

The results indicate that generating the optimum forced air flow increased the motoring losses approximately 5 pounds per square inch mean effective pressure regardless of chamber type or engine speed. With a given type of chamber, the rate of increase in friction mean effective pressure with engine speed is independent of the air-flow speed. The effect of boost pressure on the friction cannot be predicted because the friction was decreased, unchanged, or increased depending on the combustion-chamber type and design details. High compression ratio accounts for approximately 5 pounds per square inch mean effective pressure of the friction of these single-cylinder compression-ignition engines. The single-cylinder test engines used in this investigation had a much higher friction mean effective pressure than conventional aircraft engines or than the 9-cylinder, radial, compression-ignition engines tested so that performance should be compared on an indicated basis.

INTRODUCTION

The importance of air flow in mixing fuel and air in the combustion chamber of a compression-ignition engine has been recognized by investigators for a number of years and various means have been used to generate high-speed air flow in the combustion chamber.

Of the several types of combustion chamber for compression-ignition engines (reference 1), two distinct types are used in producing high-speed forced air flow. One type has a divided combustion space in which the air flow is generated by the flow of air through a fixed restriction from the cylinder into the prechamber. The other type uses a displacer that comes into operation at a predetermined point in the compression stroke and forces an orderly movement of the air from the cylinder into the combustion space.

In the study of forced and controlled air flow in the combustion chamber of an engine, most investigations (references 2 and 3) have been directed toward the determination of the optimum air-flow speed and direction and have ignored the cost of air flow in terms of friction mean effective pressure due to increased pumping losses. In the present investigation this condition has been reversed; no attempt has been made to measure the actual air speeds but the cost of generating the air flow in terms of increased friction losses has been investigated. Friction characteristics of the prechamber and displacer-type chambers have been investigated for a wide range of air-flow quantities and velocities as influenced by chamber shapes and passage sizes. The scope of the work was further broadened by determining the basic friction of the test unit with no air flow in the chamber and at a series of compression ratios covering the compression-ignition and the spark-ignition ranges.

APPARATUS AND METHODS

Two single-cylinder, 5- by 7-inch test engines were used in this work, both being water-cooled 4-stroke-cycle units identical in construction but with a small difference in crankcase bearing friction. Figure 1 shows the general arrangement of equipment for the tests. The engine bases and cylinders, of regular universal test-engine design (reference 4), were so constructed that the compression ratio could be varied at will by raising or lowering the cylinder and head. Two cylinder heads were used in this investigation, N.A.C.A. no. 7 head (fig. 2) used with and without a prechamber and N.A.C.A. no. 4 head (fig. 3) used with both a displacer piston and a conventional piston. Each had its own valve gear and each was timed for optimum 4-stroke-cycle operation. The same timing as was used in previous work on these heads (references 5 and 6) was used in the present tests. No change in valve timing was made for either head when changing from one type of combustion chamber to the other.

In all air-flow tests the procedure was the same. The combustion chamber was assembled to give the variable to be studied and the engine was motored by the dynamometer at the predetermined test speed. When all conditions had become stabilized, the dynamometer scale was read, first, after externally applying an additional load and, second, after decreasing the load. In each case the scale arm was allowed to come to rest before reading. Readings obtained in this manner varied by only 0.3 pound (0.43 pound per square inch mean effective pressure) and an average of the two readings was recorded.

Both the prechamber head and the displacer head were used in determining the effect of mass of air in motion. For the test with the prechamber head, the air-flow speed for a given engine speed was kept constant and the volume of air displaced was changed by varying the size of the chamber from 0 to 70 percent of the total clearance volume by using four different detachable auxiliary chambers. For each chamber size the total friction load was observed at 1,200, 1,500, and 1,800 r.p.m. for a compression ratio of 13.5. With the displacer piston head, the volume and speed of air displaced were changed by varying the height of the displacer that changes the crank angle at which the displacer came into action. Tests were made at 1,200, 1,500, and 1,700 r.p.m. with the displacer height varied from 7/16 inch to 1-1/2 inches.

Both cylinder heads were used in determining the effect of air-flow speed. With each head the air-flow speed was varied by changing the area of the restricting passage and keeping all other conditions constant. With air-flow speed as the variable runs were made only with both heads at 1,500 r.p.m.

In order to obtain a comparison between the optimum prechamber and the optimum displacer, the friction load of the optimum displacer was observed at 900, 1,200, 1,500, and 1,800 r.p.m. The displacer was next removed from the piston; the bolt holes in the piston crown were plugged; and, with the same compression ratio, tests were made at the same speeds. Then the prechamber head was mounted on the engine, first, with the optimum auxiliary chamber and, second, with the head converted into an integral chamber, by closing the connecting passage, and the tests were repeated for each assembly.

The prechamber head was again used first as an integral and then as an auxiliary-chamber type of combustion chamber, and the effect on the friction mean effective pressure of boosting the inlet pressure was determined. The inlet pressure was varied from 0 to 10 inches of mercury in four steps by means of an independently driven supercharger, and the friction mean effective pressure at each pressure was determined at 1,500 r.p.m.

Several supplementary tests were made to obtain complete information on friction losses. The effect of oil and water temperature, over a temperature range from 120° F. to 200° F., on the friction mean effective pressure was determined. By the use of an integral chamber and the adjustment of the compression ratio of the universal test engine (reference 4) the effect of compression ratio on friction mean effective pressure was investigated for a range of compression ratios from 6 to 20, all other conditions being constant. So that an idea of the magnitude of the air-flow losses compared to the mechanical losses might be obtained, a "breakdown" test was made which consisted of progressively removing the various operating parts of the engine and obtaining the friction of the stripped unit. The friction of the part removed was obtained by differences.

RESULTS AND DISCUSSION

When noting the reported results, consideration should be given the fact that an increase in friction mean effective pressure due to air flow was observed as an increase in the total friction of the engine. The breakdown tests, to be discussed later, show that the pumping loss at 1,500 r.p.m. was about 33 percent of the total friction; as all the increase in friction mean effective pressure should be charged to the pumping loss, an increase of 10 percent in the total friction mean effective pressure would be a 30 percent increase in the pumping loss.

Figure 4 shows that, as the proportion of the air charge forced into the prechamber was increased, the total friction mean effective pressure was increased even though the speed of the air movement was kept constant. The increase in friction mean effective pressure was proportional to the increase in the mass of air moved. These data were obtained from the motoring tests of the prechamber in which the maximum speed of the air flow was held constant by properly proportioning the area of the connecting passage to the volume of the prechamber.

In the friction tests of the displacer chamber (fig. 5) the displacer height is shown plotted against friction mean effective pressure. For the three test speeds, the friction mean effective pressure increases at an increasing rate as the displacer height is increased, showing the effect of the combination of the two variables: volume and speed.

The effect of forced air-flow speed on the friction mean effective pressure is shown in figures 6 and 7. In general, the friction mean effective pressure increases with increase in air-flow speed. This increase is a straight-line function for both the prechamber and the displacer. The speeds are computed as shown in reference 6 and no attempt has been made actually to measure them or to investigate the nature of the flow.

A comparison of the displacer chamber and the prechamber is shown in figure 8. At the higher speeds the displacer has more friction than the prechamber because of the induction throttling action of the under-sized throat. Later tests have shown that enlarging the throat area of the displacer chamber but keeping the passage area the same caused the slope of the friction curve to decrease. In either type of chamber the cost of generating the required air flow is approximately 5 pounds per square inch friction mean effective pressure.

The prechamber head was used first as an auxiliary chamber type of combustion chamber and next as an integral type of combustion chamber; the effect of boost pressure on both types of combustion chamber was determined. (See fig. 9.) As the inlet pressure was increased, a small increase in friction due to the greater density of the air forced through the connecting passage was noted for the prechamber. With the integral chamber the friction mean effective pressure is practically constant over the range of pressures tested. Later tests, using a displacer piston at 15.3 compression ratio and at 2,000 r.p.m., have shown that the friction decreases from 44 to 38 pounds per square inch mean effective pressure when the boost pressure is raised from 0 to 20 inches of mercury. Enlarging the throat area was found to decrease the friction mean effective pressure at all speeds and boost pressures tested. In every case, the supercharger was driven independently of the engine and the friction mean effective pressure measured for the engine alone.

The effect of lubricating oil and coolant temperature on friction is well known; however, in order to show the magnitude of these variables compared with changes in the pumping loss, curves are presented that show the change in friction mean effective pressure with change in lubricating oil and coolant temperature (fig. 10). These data, like all other data presented herein, are for motoring conditions; in this case, however, the trend under power was checked against that under motoring conditions, and the trends were found to be the same.

The effect of compression ratio on friction is not so universally accepted as the effect on friction of the two variables discussed in the preceding paragraph. In fact, compression ratio has been reported by other investigators (reference 7) as having no effect on the friction mean effective pressure. In the present paper, curves are presented (fig. 11) showing the effect of compression ratio over a wide range and at three speeds. The results are easily reproducible for motoring conditions and should be comparable with the other results presented in this report. The shape of the curves is significant in that they flatten out at the high compression ratios; consequently, increasing the compression ratio of a compression-ignition engine increases the friction mean effective pressure less than a corresponding increase in the compression ratio of a gasoline engine because the latter engine operates at the lower compression ratio where the rate of increase is high.

The apparently inherent and unavoidable loss due to compression-ratio increase is caused by pressure leakage and heat loss, which decrease the energy returned to the flywheel on the expansion stroke. The curves indicate that a compression-ignition engine operating at a compression ratio of 15 would have a friction mean effective pressure greater by approximately 6 pounds per square inch than a carburetor engine at a compression ratio of 7.

Table I is presented to give an idea of the friction of the different parts of the test engine used in this work. No attempt will be made to justify the method used to obtain these data and the results may be subject to minor errors: however, the authors believe the results to be indicative of the friction caused by the different parts itemized in the table. Attention is called to the high friction of this test engine. In the analysis of the other results presented this fact must be borne in mind. It should also be remembered that all power tests made with single-cylinder engines by the N.A.C.A. at Langley Field are made on this same type of test unit and, owing to its high friction, the reported brake performance is correspondingly low, therefore performance should be compared on the basis of indicated performance.

Friction characteristics of two 9-cylinder compression-ignition aircraft engines are shown in figure 12. These data have been supplied by the Bureau of Aeronautics, Navy Department. The engines are both air-cooled radial of the single poppet-valve type. Because of the low friction values and slight slope of the curves, the lower curve has been completely rechecked by test. During these tests there was no indication that combustion of the lubricating oil was decreasing the friction reading. The low friction values may be attributed in part to the use of the single large poppet valve, instead of two smaller valves, and to the complete absence of induction piping and blower. The test results are presented as the only available information regarding the friction characteristics of aircraft compression-ignition engines.

CONCLUSIONS

1. The increase in friction mean effective pressure due to forced air flow is directly proportional to the increase in volume and speed of the displaced air.

2. With a given type of chamber, the rate of increase in friction mean effective pressure with engine speed is independent of the air-flow speed.

3. The effect of boost pressure on friction cannot be predicted because the friction was decreased, unchanged, or increased with increase in boost pressure, depending on the combustion-chamber type and design details.

4. For optimum conditions, the forced air-flow friction amounted to approximately 5 pounds per square inch mean effective pressure regardless of the engine speed or the type of combustion chamber.

5. High compression ratio accounts for approximately 5 pounds per square inch mean effective pressure of the friction of these single-cylinder compression-ignition engines.

6. The friction mean effective pressure of this type of single-cylinder engine, even without air flow and at a compression ratio of 5.5, is much higher than that of conventional aircraft engines or the 9-cylinder radial compression-ignition engines tested and performance should therefore be compared on an indicated basis.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., July 9, 1936.

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TABLE I

Breakdown Tests of Test Unit 1

(Prechamber cylinder head as integral chamber; compression ratio, 13.5; oil temperature - out, 180° F.; water temperature - out 170° F.; engine speed, 1,500 r.p.m.)

Engine part	f.m.e.p.	Percentage of total losses
	lb./sq.in.	
Pumping losses	12.1	32.8
Piston, rings and rod	11.1	30.0
Main bearings	8.0	21.7
Valves and valve gear	2.7	7.3
Water pump	1.5	4.0
Oil pump	.5	1.4
Fuel pump (Throttle closed)	0	0
Fuel pump (Injecting full load)	1.0	2.8
Total losses, integral chamber	36.9	100.0
(Total losses 70 percent prechamber)	(41.0)	

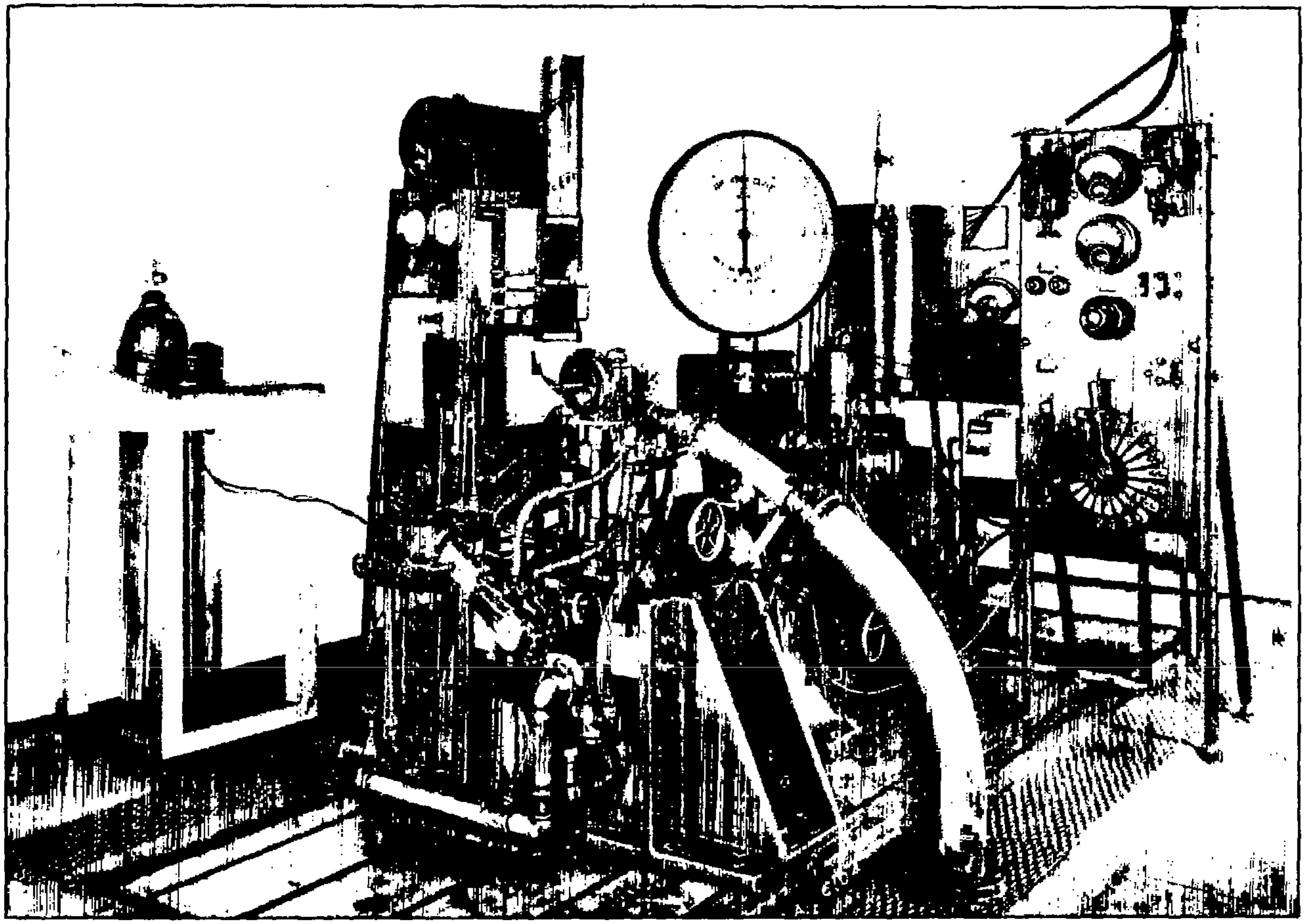

Figure 1.- Single-cylinder engine and test equipment.

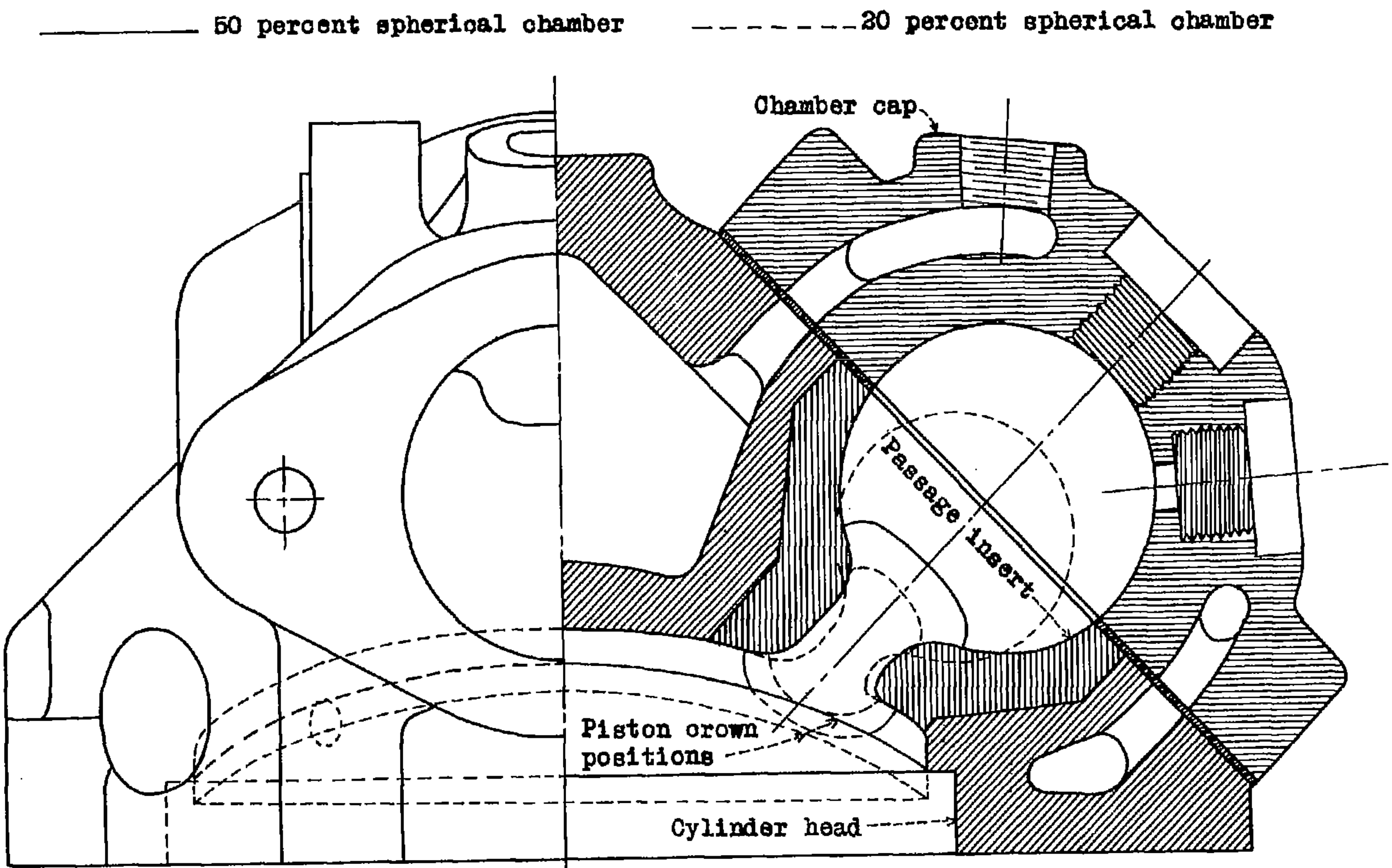

Figure 2.- Prechamber combustion chamber.

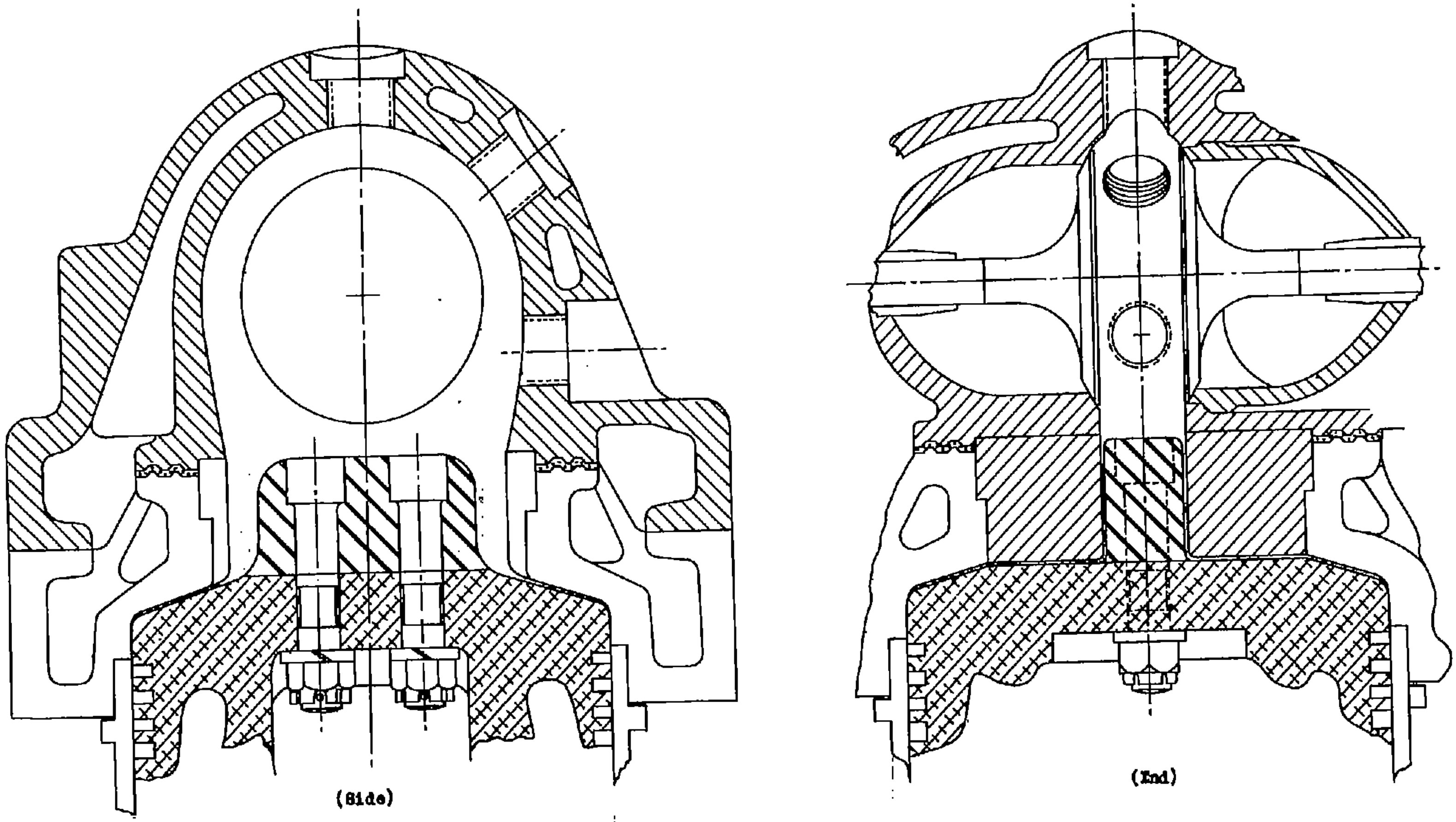

Figure 3.- Displacer-piston combustion chamber.

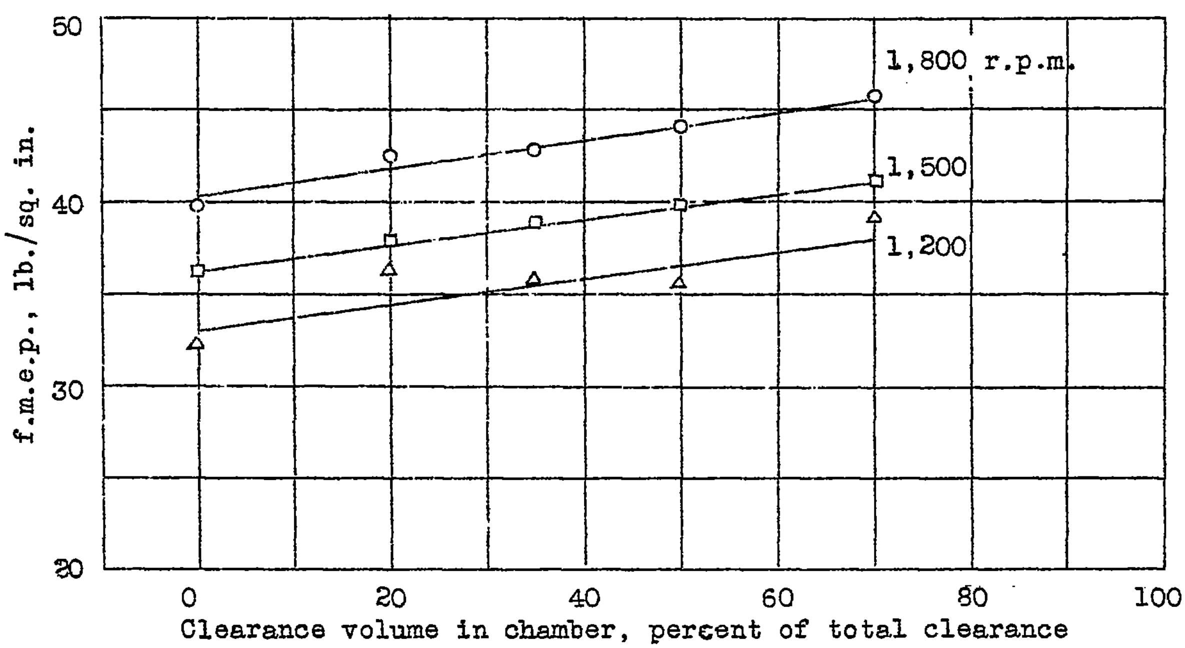

Figure 4.- Effect of prechamber size on f.m.e.p. Test unit 1; prechamber cylinder head; compression ratio, 13.5; oil temperature - out, 180° F.; water temperature - out, 170° F.

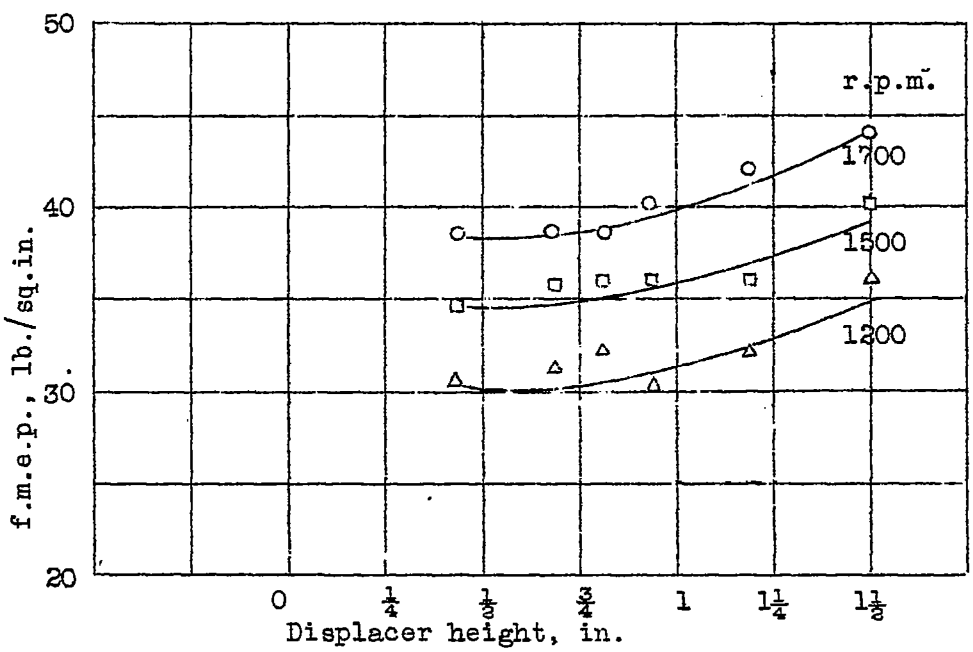

Figure 5.- Effect of displacer height on f.m.e.p. Test unit 2; displacer combustion chamber; compression ratio, 13 to 15.7; oil temperature - out, 165° F., water temperature - out, 170° F.

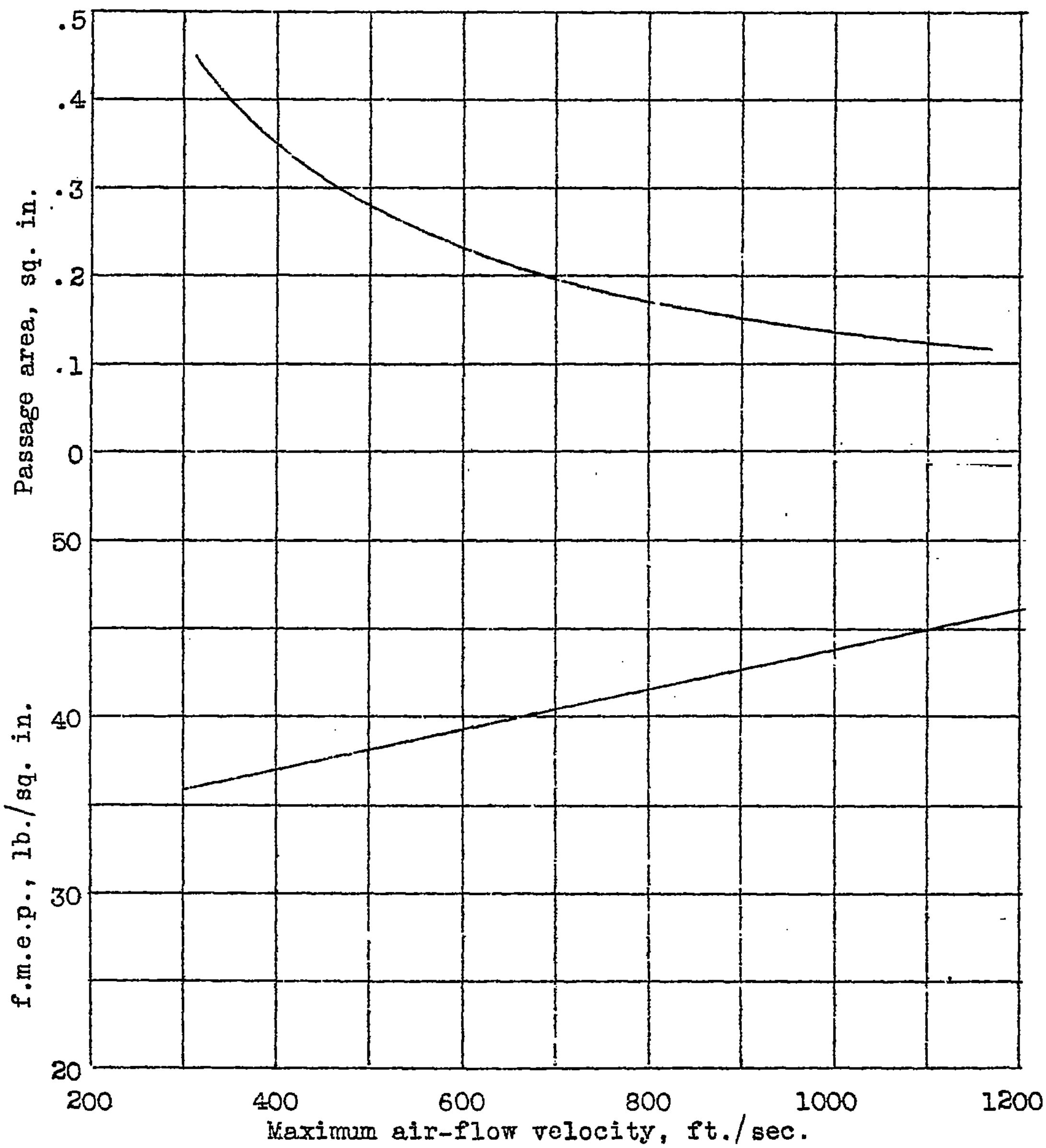

Figure 6.- Effect of maximum air-flow speed on f.m.e.p. Test unit 1; prechamber cylinder head; 50 percent clearance in prechamber; compression ratio, 13.5; oil temperature - out, 140° F.; water temperature - out, 170° F.; engine speed, 1,500 r.p.m.

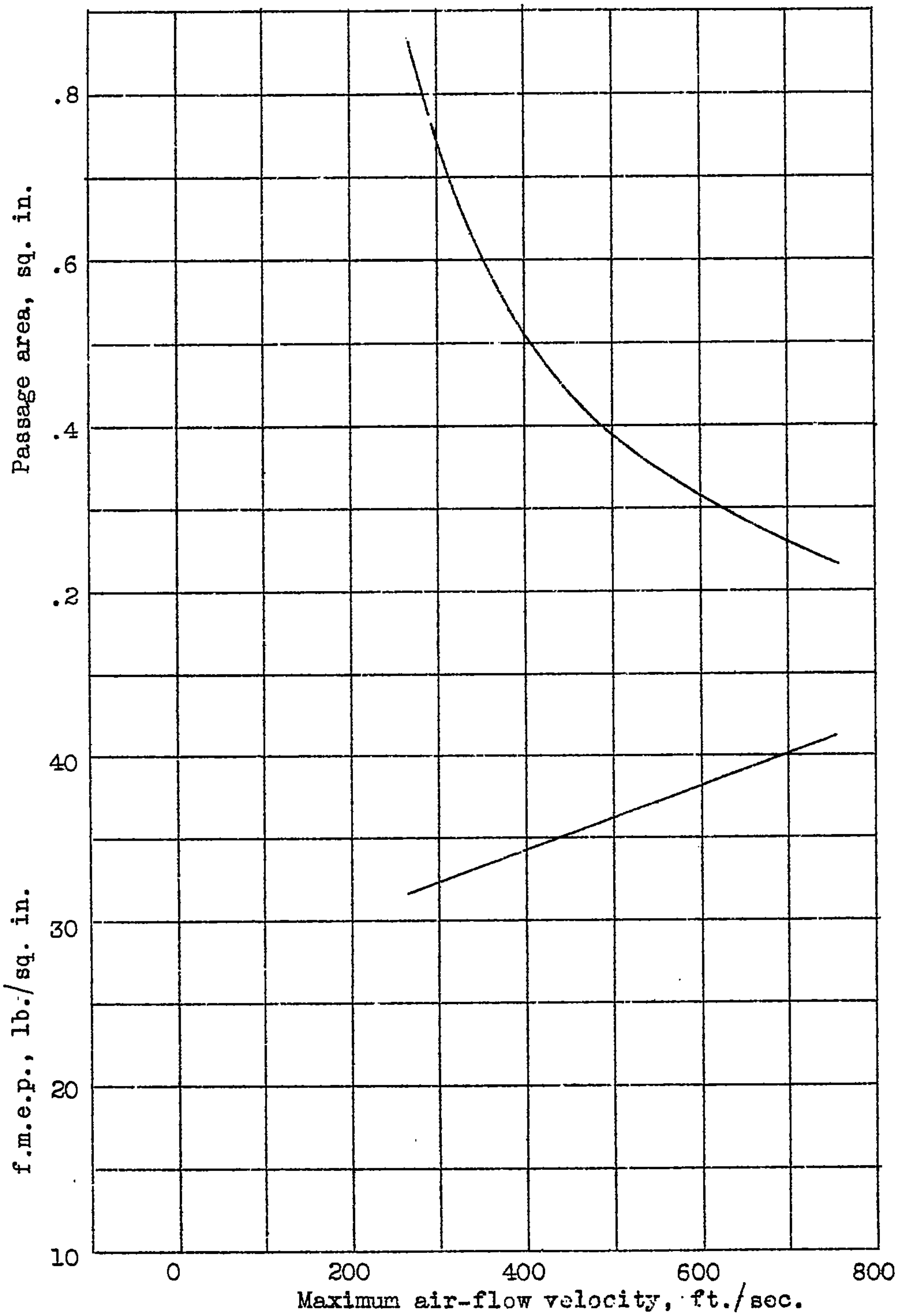

Figure 7.- Effect of maximum air-flow speed on f.m.e.p. Test unit 2; displacer combustion chamber; compression ratio, 15.3; oil temperature - out, 165° F.; water temperature - out, 170° F.; engine speed, 1,500 r.p.m.

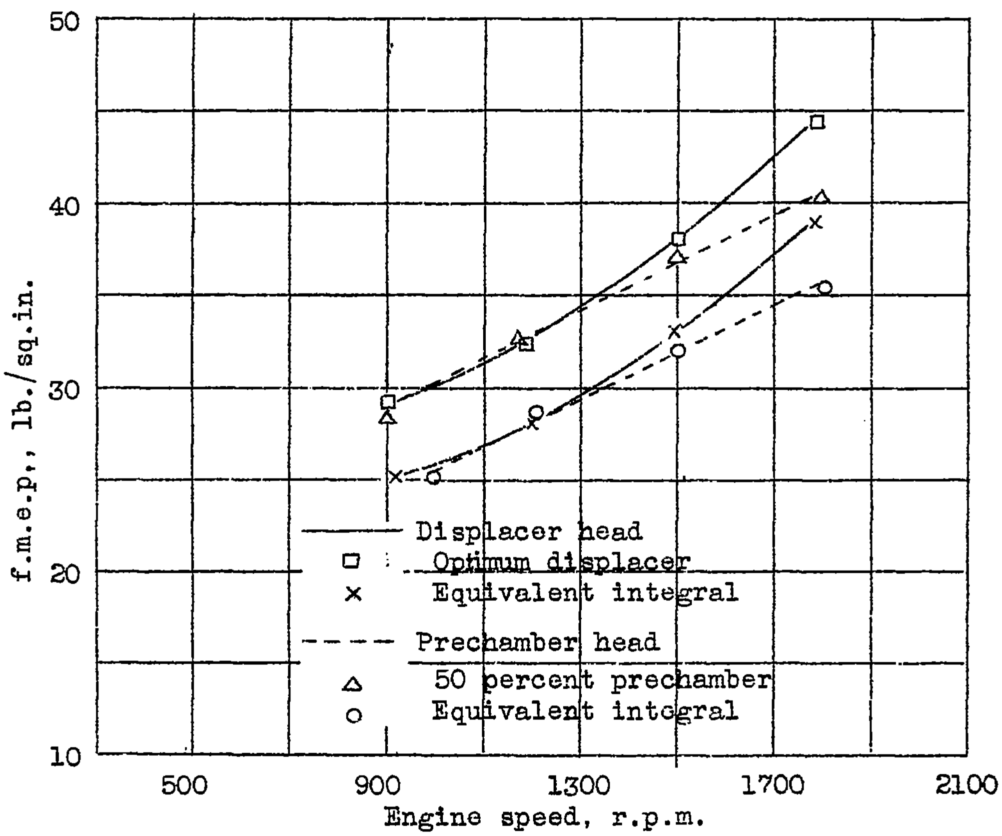

Figure 8.- Comparison of prechamber and integral chamber with displacer combustion chamber and integral chamber. Test unit 1; compression ratio, 15.3; oil temperature - out, 180° F.; water temperature-out, 170° F.

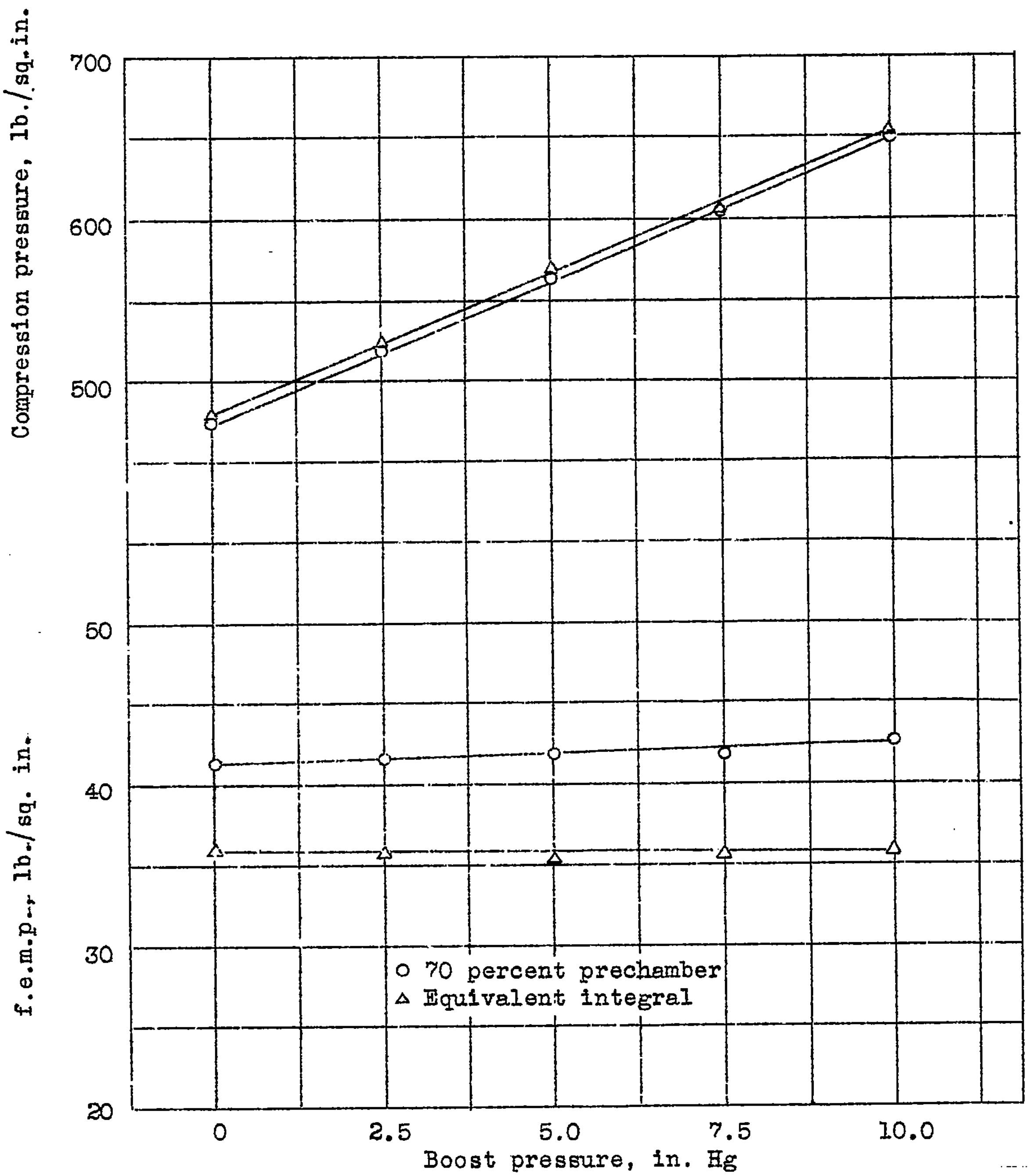

Figure 9.- Effect of boost pressure on f.m.e.p. using both integral and 70-percent chamber. Test unit 1, prechamber cylinder head; compression ratio, 13.5, oil temperature - out, 180° F., water temperature - out, 170° F., engine speed, 1,500 r.p.m.

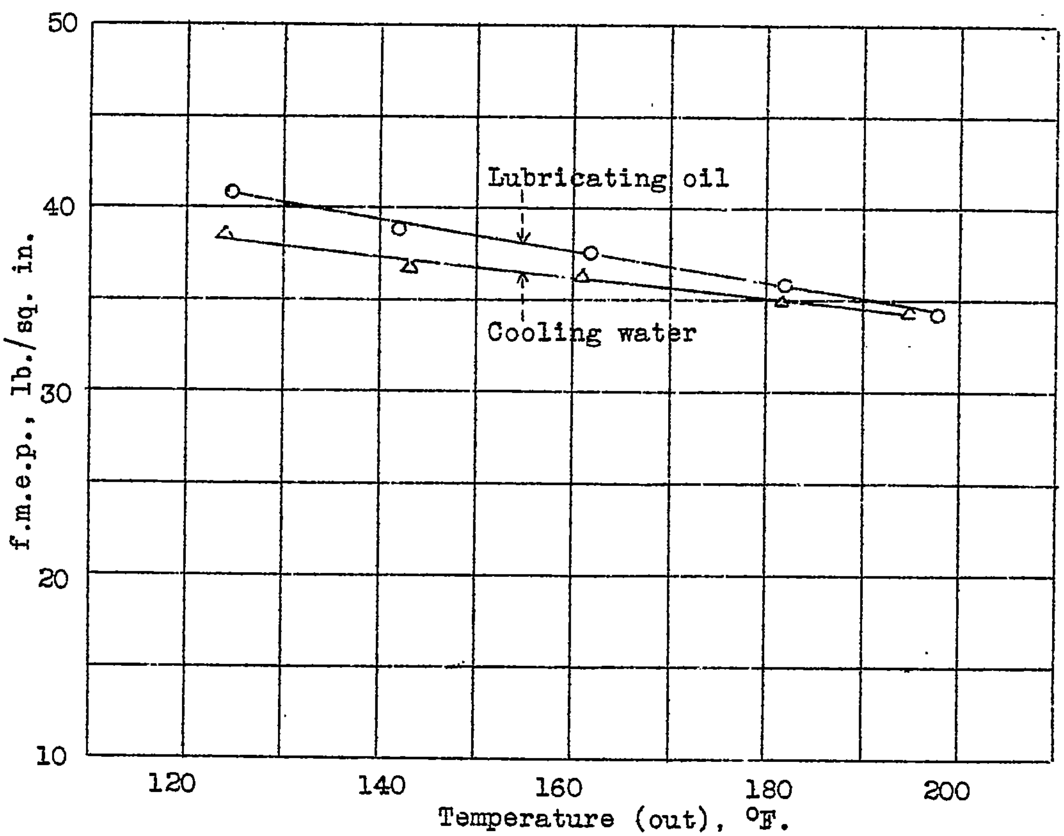

Figure 10.- Effect of lubricating oil and cooling-water temperature on f.m.e.p. Test unit 1; prechamber cylinder head, integral chamber; compression ratio, 13.5; standard oil temperature- out, 180° F.; standard water temperature - out, 170° F., engine speed, 1,500 r.p.m.

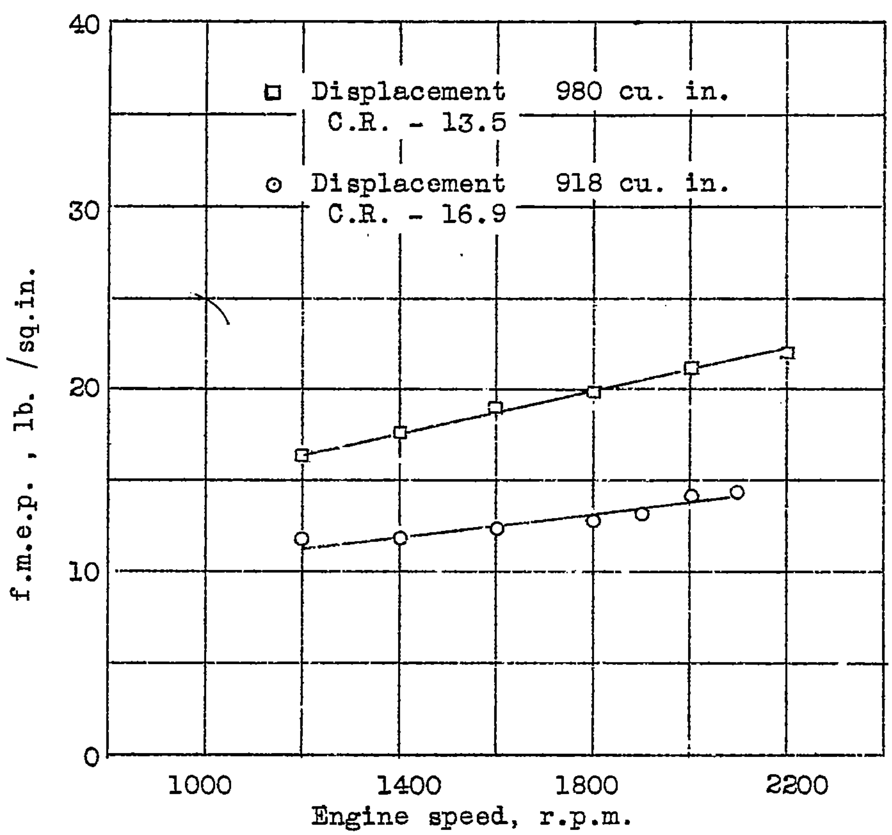

Figure 12.- Effect of speed on f.m.e.p. for 9-cylinder, radial, air-cooled, 4-stroke cycle, single-valve engines. Lubricating oil temperature - out , 150° F.